Part 2

# Standard ASME B16.5 Flanges: Bolt Tightening and Target Loads

Using the approach outlined here, piping engineers can select the most appropriate bolt-tightening technique based on pressure class and flange size, and calculate the required torque or tool pressure if hydraulic tensioning tools are used

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IN BRIEF



**FIGURE 1.** Bolted metallic flanges are a common sight on the vessels and piping systems at production plants

In the majority of industrial plants, flanged connections are used in metallic piping systems (Figure 1). The applied flanges comply with ASME B16.5 [1]. Achieving a proper preload in the bolts of a gasketed, bolted-flanged joint during joint assembly is considered crucial for its optimized performance. To realize this, various bolt-tightening techniques are available for the flange fitter. Applying appropriate tightening depends on bolt size and pressure class of the flange proper. In addition to other aspects with regard to bolted joints and their sealing performance, the appropriate and prudent choice must be made of the tightening tool and the bolt load that must be realized during assembly. This article is intended as a practical guideline for piping mechanical engineers and focuses in particular on two specific aspects, namely:

- A selection chart that provides a demarcation between various tightening methods of bolts in standard flange connections
- An algorithm that enables the engineer involved in the design to determine the target bolt load from which the required torque or tension force can be derived

The approach described here offers the piping mechanical engineer the opportunity to select and differentiate among bolt tightening techniques for extensive piping systems based on pressure class and flange diameters. In addition, the option is offered to determine the required torque or the tool pressures if hydraulic tensioning tools are used.

Using this approach in practice, it has been shown that costs can be saved on flange assembly work without compromising safety and reliability.

## Bolt tightening selection chart

The selection chart shown in Table 1 shows the demarcation between three different bolt-tightening techniques for standard ASME B16.5 flanges. One can see that as the pressure class increases, the tightening method becomes more critical.

Table 2 is a further elaboration of Table 1, and includes the number of bolts (#) and the bolt size (size), with a color marking indicating the recommended tool type.

## Background and scope

The selection chart (Tables 1 and 2) was developed in the late 1980s at the request of the maintenance engineering department of a large exploration and production company located in the Netherlands. The core business of that company is exploring for and production of oil and gas, both onshore and offshore. The scope of the selection chart had to extend to piping pressure classes 150 to 2,500. Applicable flange materials are, successively: carbon steel, austenitic stainless steel and austenitic-ferritic (duplex) stainless steel. The chart is applicable for temperatures up to 200°C for all piping class materials. The concerned stud bolt materials are provided in Table 3.

## Specifics

Conventional hand-operated (manual) torque wrenches are specified for bolt sizes up to 1 in. Those wrenches usually deliver a maximum torque between 700 and 1,000 Nm. The bolt load accuracy for this method lies between 25 and 35%.

A bolt load accuracy of approximately 50% applies to the mechanical or hydraulic torque wrench, while an accuracy of 90% is achievable for the hydraulic bolt tensioning method.

Critical applications require special consideration. Proper tightening procedures must be strictly followed.

In some cases where hydraulic bolt tensioning is recommended, it may turn out that the use of hydraulic bolt tensioning equipment is not possible due to lack of space to install the equipment, for instance, in case of close assemblies of two valves.

In those cases, hydraulic torque wrenches are recommended provided that in addition to the use of hydraulic torque wrenches, the equal bolt stress distribution shall be controlled by means of proper bolt elongation measurements (for example, ultrasonic measuring devices).

Also, mechanical torque wrenches or hydraulic torque wrenches may be used instead of hydraulic bolt tensioning equip-

**TABLE 1. DEMARCATION BETWEEN BOLT TIGHTENING TECHNIQUES FOR FLANGES CONFORMING TO ASME B16.5**

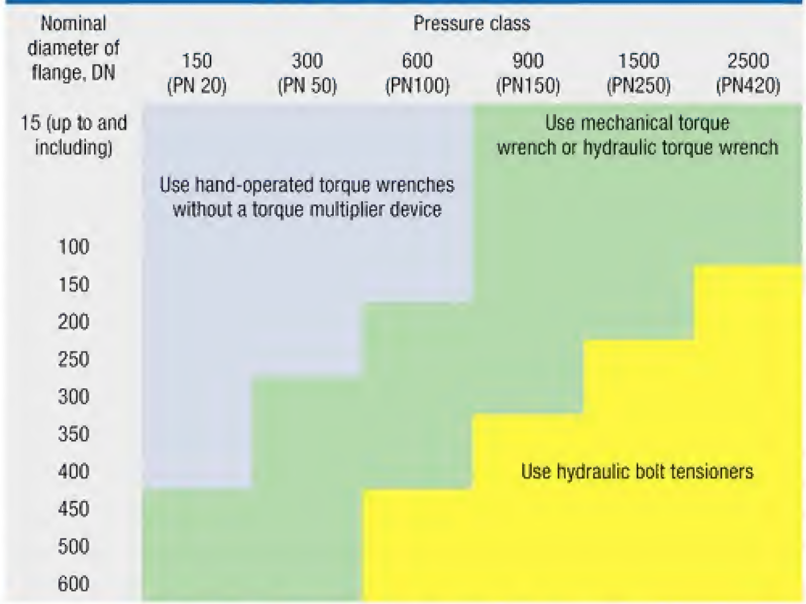


**TABLE 2. ASME B16.5 FLANGE BOLTING INFORMATION**

| Flange size (NPS - DN) | Class 150 # - Size | Class 300 # - Size | Class 600 # - Size | Class 900 # - Size | Class 1500 # - Size | Class 2500 # - Size |
|---|---|---|---|---|---|---|
| ½- 15 | 4 - ½ | 4 - ½ | 4 - ½ | 4 - ¾ | 4 - ¾ | 4 - ¾ |
| ¾- 20 | 4 - ½ | 4 - 5/8 | 4 - 5/8 | 4 - ¾ | 4 - ¾ | 4 - ¾ |
| 1 - 25 | 4 - ½ | 4 - 5/8 | 4 - 5/8 | 4 - 7/8 | 4 - 7/8 | 4 - 7/8 |
| 1½ - 40 | 4 - ½ | 4 - ¾ | 4 - ¾ | 4 - 1 | 4 - 1 | 4 - 1 1/8 |
| 2 - 50 | 4 - 5/8 | 8 - 5/8 | 8 - 5/8 | 8 - 7/8 | 8 - 7/8 | 8 - 1 |
| 3 - 80 | 4 - 5/8 | 8 - ¾ | 8 - ¾ | 8 - 7/8 | 8 - 1 1/8 | 8 - 1¼ |
| 4 - 100 | 8 - 5/8 | 8 - ¾ | 8 - 7/8 | 8 - 1 1/8 | 8 - 1¼ | 8 - 1½ |
| 6 - 150 | 8 - ¾ | 12 - ¾ | 12 - 1 | 12 - 1 1/8 | 12 - 1 3/8 | 8 - 2 |
| 8 - 200 | 8 - ¾ | 12 - 7/8 | 12 - 1 1/8 | 12 - 1 3/8 | 12 - 1 5/8 | 12 - 2 |
| 10 - 250 | 12 - 7/8 | 16 - 1 | 16 - 1¼ | 16 - 1 3/8 | 12 - 1 7/8 | 12 - 2½ |
| 12 - 300 | 12 - 7/8 | 16 - 1 1/8 | 20 - 1¼ | 20 - 1 3/8 | 16 - 2 | 12 - 2¾ |
| 14 - 350 | 12 - 1 | 20 - 1 1/8 | 20 - 1 3/8 | 20 - 1½ | 16 - 2¼ | - |
| 16 - 400 | 16 - 1 | 20 - 1¼ | 20 - 1½ | 20 - 1 5/8 | 16 - 2½ | - |
| 18 - 450 | 16 - 1 1/8 | 24 - 1¼ | 20 - 1 5/8 | 20 - 1 7/8 | 16 - 2¾ | - |
| 20 - 500 | 20 - 1 1/8 | 24 - 1¼ | 24 - 1 5/8 | 20 - 2 | 16 - 3 | - |
| 24 - 600 | 20 - 1¼ | 24 - 1½ | 24 - 1 7/8 | 20 - 2½ | 16 - 3½ | - |

The colors refer to the recommended tightening tools given in Table 1

Key:

- Use hand-operated torque wrenches without a torque multiplier device
- Use mechanical or hydraulic torque wrench
- Use hydraulic bolt tensioners

| TABLE 3. SPECIFIED BOLTING MATERIALS TO ASTM | | | | | | |
|---|---|---|---|---|---|---|
| Diameter | Studbolts | | | Nuts | | |
| [in.] | Non-sour service | Sour service | Low-temperature service | Non-sour service | Sour service | Low-temp. service |
| ≤ 1 ½ | A 193-GR.B7 | A 193-GR.B7M | A 320-GR. L7 | A 194-GR.2H | A 194-GR.2HM | A 194-GR.7 |
| 1 ⅝ – 2 ½ | A 320-GR.L7 | A 320-GR.L7M | | A 194-GR.7 | A 194-GR.7M | |
| > 2 ½ | A 320-GR.L43 | | A 320-GR. L43 | | | |

| TABLE 4. EQUATIONS FOR CALCULATING TOTAL REQUIRED BOLT LOAD | | |
|---|---|---|
| Total required bolt load *(W)* including internal pressure *(P)* and external loads *(F + M)* | Total required bolt load (W) for hydrostatic test pressure $(P_T)$ without external loads | Total required bolt load (W) for gasket seating (GS) |
| $W_{(P+F+M)}$ | $W_{(PT)}$ | $W_{(GS)}$ |
| $2\pi GP(G/4 + 2bm)$ | $\pi GP_T(G/4 + 2bm)$ | $\pi bGy$ |
| Required bolt load per bolt | Required bolt load per bolt | Required bolt load per bolt |
| $W_{(P+F+M)} / n_B$ | $W_{(PT)} / n_B$ | $W_{(GS)} / n_B$ |

Notation
$W$ = total required bolt load, N
$G$ = diameter at location of gasket load reaction*, mm
$P$ = internal design pressure, MPa
$P_T$ = hydrostatic test pressure, MPa
$b$ = effective gasket or joint-contact-surface seating width*, mm
$m$ = gasket factor, unitless
$y$ = gasket or joint-contact-surface unit seating load, MPa
$n_B$ = number of bolts, unitless

*Note: Consult Table 2-5.2 of Ref. 2 for obtaining effective gasket width and diameter at location of gasket load reaction.

| TABLE 5. DATA FOR THREE FLANGES IN THE EXAMPLE | | | |
|---|---|---|---|
| Nominal diameter | NB 150 | NB 300 | NB 500 |
| Class | 600 | 600 | 600 |
| Bolt size | 1 in. | 1¼ in. | 15/8 in. |
| Number of bolts | 12 | 20 | 24 |
| Flange material | A105 | A105 | A105 |
| Bolting material | A 193-GR.B7 | A 193-GR.B7 | A 320-GR.L7 |
| Gasket type [4] | Spiral wound | Spiral wound | Spiral wound |
| Gasket dimensions, mm [4] | ID 174.8; OD 209.6 | ID 327.2; OD 374.7 | ID 520.7 / OD 577.9 |
| Gasket bead, mm | 1.5 | 1.5 | 1.5 |
| Gasket width, $N$, mm | 17.4 - 1.5 = 15.9 | 23.75 - 1.5 = 22.25 | 28.6 -1.5 = 27.1 |
| Basic gasket seating width, mm | $N/2 = 7.95$ | $N/2 = 11.125$ | $N/2 = 13.55$ |
| Effective gasket seating width, mm | $b = 2.52\sqrt{7.95} = 7.11$ | $b = 2.52\sqrt{11.125} = 8.4$ | $b = 2.52\sqrt{13.55} = 9.28$ |
| Diameter at location of gasket load reaction, mm | $G = 192.38$ | $G = 354.9$ | $G = 556.34$ |
| Gasket factor, dimensionless | $m = 2.5$ | $m = 2.5$ | $m = 2.5$ |
| Gasket seating stress, MPa | $y = 69$ | $y = 69$ | $y = 69$ |
| Grip length bolt, mm | - | - | $C = 196.3$ |
| Nominal bolt diameter, mm | - | - | $D = 41.275$ |

ment in those cases where decreased operating pressures do not require high bolt stresses to maintain a reliable flange connection. In those cases, the responsible party shall provide for calculations to demonstrate the integrity of the flange connection. It is also emphasized that too low bolt stresses may lead to unstable flange connections.

## Algorithm to calculate target bolt load

When determining the target bolt load, *W*, the following loads or combinations of loads should be considered:

***Internal (design) pressure.*** External loads must also be taken into account.

***Hydrostatic test pressure.*** Flanged joints may be subjected to system hydrostatic tests at a pressure of 1.5 times the 38°C rating. However, the actual hydrostatic test pressure can be lower. External loads can be omitted with this condition.

***Imposed external loads.*** If the magnitude of external loads is unknown, then this load may be approximated by doubling the internal (design) pressure. The background for this arises from the fact that the ratio between circumferential stress and longitudinal stress in the adjacent connected pipe amounts to a factor of two. This is equivalent to doubling the force from the adjoining cylindrical part.

***Gasket seating.*** Before a tight joint can be obtained, it is necessary to seat the gasket or joint-contact surface properly by applying a minimum initial load.

Note that ASME BPVC Section VIII-Division 1 "Mandatory Appendix 2 Rules for Bolted Flange Connections with Ring-Type Gaskets," [2] is applied as much as possible.

Table 4 lists the expressions used for calculating the total required bolt load, depending on the loads mentioned above.

## Determining target torque

A common method for converting the target-bolt-assembly load into the target torque is the use of the following equation [3]:

$$T = (K \times D \times W)/1{,}000 \qquad (1)$$

Where:
$D$ = nominal diameter of the bolt, mm
$W$ = target bolt load: max $[W_{(P+F+M)}/n_B ; W_{(PT)}/n_B ; W_{(GS)}/n_B]$, N
$K$ = nut factor, to be set at 0.2 for non-coated bolts
$T$ = target torque, Nm

## Determining tool pressure

The following relations are used to determine the tool pressure required to achieve the target bolt load:

$$\text{Tool pressure} = (\text{Target bolt load} \times LTF)/\text{Tool hydraulic pressure area (HPA)} \quad (2)$$

Where:
Target bolt load = required residual bolt load = max $[W_{(P+F+M)}\ ;\ W_{(PT)}\ ;\ W_{(GS)}]$, N
$LTF$ = load transfer factor = $1.01 + (D/C) \mapsto$ Minimum 1.1
$D$ = nominal bolt diameter, mm
$C$ = clamping or joint grip length, mm

Tensioning permits the simultaneous tightening of multiple bolts; the tools are connected in sequence via a high-pressure hose assembly to a single pump unit. This ensures that each tool develops the exact same load and provides a uniform clamping force across the joint. This is especially important for critical piping requiring even gasket compression to effect a seal. Hence, account must be taken of the simultaneous hydraulic tightening of a number of bolts (studs) to determine the required tool pressure. The pre-tension per bolt is the target bolt load or required residual bolt load divided by the number of bolts, $n_B$.

## Worked example

This example is based on the assumption that there are three different flange pairs, the bolts of which must be tightened with an appropriate method. The three flanges, together with their relevant flange, gasket and bolt data, are summarized in Table 5. Note that the color corresponds to the recommended tightening techniques listed in the selection chart (Table 1). The system pressures and temperatures for class 600 piping system (Group 1.1 Materials [*1*]) are given in Table 6.

With this information, and the equations provided above, a summary of the calculations for the three flanges is collected in Table 7. The target assembly loads and their respective hydraulic tool pressure is shown in bold font.

## Discussion

A comparison between the target torque values calculated in the example and the target torque values as stated in Table 1 of Ref. 3 is presented in Table 8. The large difference between the values calculated in the example and those according to Table 1 of Ref. 3 is due to the fact that a specific flange load situation (pressure, temperature, and so on) has been used in the ex-

**TABLE 6. SYSTEM PRESSURES AND TEMPERATURES FOR CLASS 600 PIPING SYSTEM (GROUP 1.1 MATERIALS [1])**

| Design temperature, °C | Design pressure, bar / MPa | Hydrostatic test pressure, bar / MPa * | Rated pressure at 200°C, bar / MPa [1] | Rated pressure at 38°C, bar / MPa [1] |
|---|---|---|---|---|
| 200 | 65 / 6.5 | 107.9 / 10.79 | 87.6 / 8.76 | 102.1 / 10.21 |

*Note: as per applicable design code

**TABLE 7. CALCULATION FORM FOR WORKED EXAMPLE**

| Nominal flange diameter | NB 150 (NPS 6 in.) | NB 300 (NPS 12 in.) | NB 500 (NPS 20 in.) |
|---|---|---|---|
| $W_{(P+F+M)}$ | | | |
| $2\pi GP\,(G/4 + 2bm)$ | 657,193 N | 1,894,776 N | 4,214,464 N |
| $W_{(PT)}$ | | | |
| $\pi GP_T\,(G/4 + 2bm)$ | 545,470 N | 1,572,664 N | 3,498,005 N |
| $W_{(GS)}$ | | | |
| $\pi bGy$ | 296,503 N | 646,226 N | 1,119,147 N |
| $W_{bolt}$ | 657,193/12 = 54,766 N | 1,894,776/20 = 94,739 N | 4,214,464/24 = 175,603 N |
| $T = K\,D\,W/1{,}000$ | 0.2× 25.4 × 54,766 /1,000 | 0.2 × 31.75 × 94,739/1,000 | |
| Target torque | **278 Nm** | **602 Nm** | Not applicable |
| $LTF = 1.01 + (D/C)$ | | | 1.01 + (41.275/196.3) = 1.22 |
| $W_{bolt} \times LTF$ | | | 175,603 x 1.22 = **214,282 N** |
| HPA of selected tool | | | 3,674 mm$^2$ (*) |
| Tool pressure | | | 58.32 MPa = **583.2 bars** |

*Note: Hydratight tool reference number AJ4

**TABLE 8. COMPARISON OF CALCULATION VERSUS TABLE 1 OF REF. 3**

| Flange identification | Bolt size | Target torque of example, Nm | Associated bolt stress, MPa | Target torque of Table 1 of Ref. 3, Nm | Associated bolt stress, MPa |
|---|---|---|---|---|---|
| Flange: NB 150 (NPS 6 in.) | 1 in. - UNC | 278 | 154 | 677.9 | 345 |
| Flange: NB 300 (NPS 12 in.) | 1¼ in. - 8UN | 602 | 158 | 1423.6 | 345 |

**TABLE 9. THE PROBABLE BOLT STRESS, $\sigma_{BOLT}$, DEVELOPED MANUALLY, AS CITED IN THREE CODES**

| Source | ASME BPVC VIII Div.1; Appendix S | EN 13445-3-Annex G | Rules for pressure vessels (RfPV); D 0701 |
|---|---|---|---|
| Bolt size: 1 in.- UNC | $\sigma_{bolt}$ = 310 MPa | $\sigma_{bolt}$ = 198 MPa | $\sigma_{bolt}$ = 268 MPa |
| Bolt size: 1¼ in. - 8UN | $\sigma_{bolt}$ = 277 MPa | $\sigma_{bolt}$ = 177 MPa | $\sigma_{bolt}$ = 240 MPa |

ample, whereas for the values according to Ref. 3, use is made of a generic situation where a target pre-stress of 345 MPa (approximately 50% yield strength of bolt material) is taken as the starting point.

Another very important item in bolting design is the question of whether the necessary bolt stress is actually realized, and what special means of tightening, if any, must be employed. Most joints are tightened manually by ordinary wrenching, and it is advantageous to have designs that require no more than this. Some pitfalls must be avoided, however. The probable bolt stress developed manually, when using standard wrenches, is cited in a number of recognized codes and in the specialized literature. An overview is found in Table 9. There one can see that there is a considerable spread in probable bolt stresses for the bolt dimensions from the example. The calculated bolt stresses of 154 MPa (for the 1 in. bolt size used in the NB 150 flange) and 158 MPa (for the 1 ¼ in. bolt size used in the NB 300 flange), are significantly lower than those that can be achieved by means of simple conventional manual tightening tools (for instance, wrenches). ■

*Edited by Gerald Ondrey*

## References


1. ASME B16.5 - 2017; Pipe Flanges and Flanged Fittings NPS 1/2 Through NPS 24.
2. ASME BPVC VIII Div.1- 2017; Appendix 2.
3. ASME PCC-1- 2013; Appendix K.
4. ASME B 16.20 - 2012; Metallic Gaskets for Pipe Flanges Ring-Joint, Spiral-Wound, and Jacketed.


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